FIELD TESTING OF GAS TURBINE DRIVEN CENTRIFUGAL COMPRESSOR PACKAGES—TEST PROCEDURES AND MEASUREMENT UNCERTAINTIES

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ABSTRACT

Field testing of gas turbine compressor packages requires the accurate determination of efficiency, flow, head, power, and fuel flow in sometimes less than ideal working environments. Nonetheless, field test results have significant implications for the compressor and gas turbine manufacturers and their customers. Economic pressures demand that the performance and efficiency of an installation are verified. Thus, an accurate determination of the field performance is of vital interest, not only to the user of compressors and gas turbines, but also to their manufacturers.

Discussed herein is the field testing of gas turbine driven compressors and the measurement uncertainties one can expect when following appropriate test guidelines. Also mentioned are common problems and considerations regarding field testing.

INTRODUCTION

Field testing of gas turbine compressor packages is becoming increasingly frequent because economic pressures demand that an installation's efficiency, capacity, head, power, and fuel flow be verified to assure a project's return on investment. However, during the field tests, an accurate determination of the gas compressor and gas turbine performance is often difficult because of less than ideal working environments. Nonetheless, field test results may have significant financial implications for the compressor and gas turbine manufacturers and their customers. Thus, for the end user and the manufacturer, an accurate determination of the compressor field performance and measurement uncertainty is of vital interest. Discussed herein is the field testing of gas turbine driven compressors and the typical measurement uncertainties one can expect. The authors suggest that a properly planned and conducted field test may sometimes be a more economical and practical solution than a PTC 10 Class I [1] factory test.

A compressor set field testing procedure that reduces measurement inaccuracies and maintains cost efficiency is outlined herein. Special attention is given to the preparation and organization of the test. Also addressed are the issues of necessary instrumentation and the interpretation of test data. Applicable test codes and their relevance for field testing are reviewed. The appropriate use of portable computers, which have introduced new powerful tools for analysis, data acquisition, and data reduction even in remote locations, is also discussed.

Finally, the equations governing the compressor and gas turbine performance uncertainties are rigorously derived and results are compared to typical field test data. Test parameters that correlate to the most significant influence on the performance uncertainties are identified and suggestions are provided on how to optimize their accuracy. Equations of state are used to calculate necessary gas parameters from the gas composition. The effect of different equations of state on the calculated performance is discussed. Results show that compressor efficiency uncertainties can be unacceptably high when basic rules for accurate testing are violated. However, by following some simple measurement rules and maintaining commonality of the gas equations of state, the overall compressor package performance measurement uncertainty can be limited and meaningful results can be achieved.

The authors emphasize that, besides the issues mentioned previously, field tests can provide users with a valuable basis for trending. They also provide equipment manufacturers with information complementary to the data collected during factory testing. Furthermore, examined herein are some questions often encountered in preparing and conducting field performance tests, including:

- Planning and administration of the field test
- Measurements and their relative importance
- Equations of state
- Comparison of factory test to field test

- Test uncertainties
- Evaluation of test results

Tests on other subjects such as vibration, emissions, and control systems are not covered within the scope of this study. The tight focus on performance is justified by very few published references to the methods applied and the lessons learned in actual field tests. Indeed, field testing has received very limited attention, considering the broad scope and depth of available technical literature on turbomachinery.

PLANNING FOR FIELD TESTS

The challenges of field tests arise not so much in applying the laws of physics and engineering that govern the behavior of turbomachinery, but in the depth of preparation and the organization of the necessary tools, conditions, and personnel required to conduct the tests and analyze the results.

Because each field installation is unique, location-specific test agendas must be prepared to supplement standard test specifications. Typical field installations are shown in Figures 1 and 2.

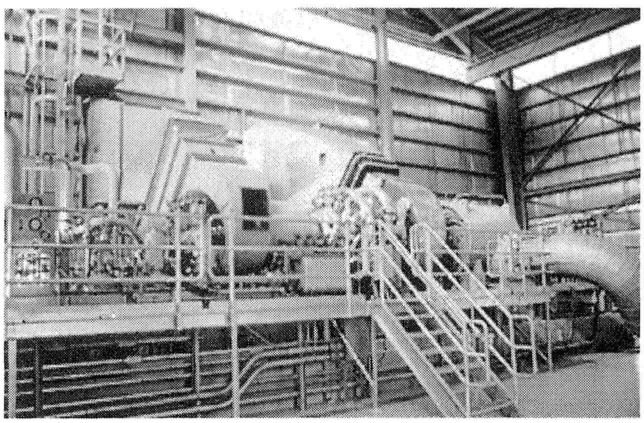

Figure 1. Gas Turbine Driven Pipeline Compressor Installation.

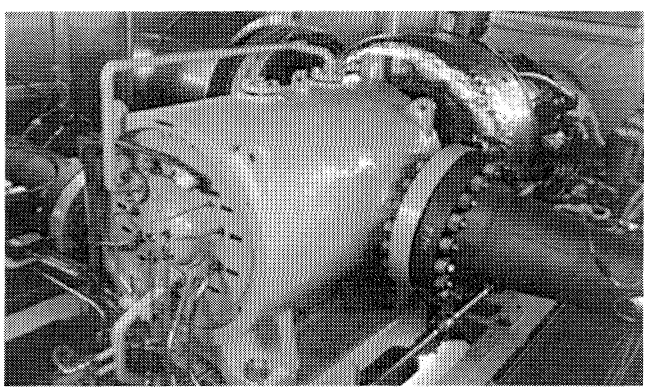

Figure 2. Field Test on a Gas Compressor Driven by a Gas Turbine.

The American Society of Mechanical Engineers (ASME), the International Organization for Standardization (ISO), and the Verein Deutscher Ingenieure (VDI) have issued specifications covering thermodynamic calculation methods, instruments, site preparation, and the reporting of turbomachinery test results in various degrees of detail. ASME Power Test Codes (PTC) 10 [1], 19.1 [2], and 22 [3], ISO 2314 [4], VDI 2045 [5], and 2048 [6] are examples of these standards.

Compliance with such specifications is a relatively easy matter in a factory environment where facilities are designed specifically for testing; qualified support personnel, instrumentation, and calibration laboratories are available; and real-time online computers routinely monitor test progress. This usually is not the case at actual installation sites designed for commercial operation of turbomachinery. Site performance tests generally require concerted planning and execution, including development of a unique test agenda prepared jointly by the manufacturer and the equipment end user. Such an agenda should communicate the field conditions and equipment layout, list the instruments to be used and their location, describe the method of operation and the pressure and temperature limits of the facility, and specify any deviations from normal operation that may be necessary to conduct the test. The items of such a test agenda can and should be discussed in a very early stage of the project.

Preparations also include discussions on available operating conditions and operational limitations. In many cases, a specified operating point can only be maintained for a limited period of time (i.e., because the pipeline operation depends on the tested package), or at fixed ambient conditions (i.e., if the necessary gas turbine power is only available on cold days).

The selection and calibration of the test instrumentation are extremely important. Generally, the instruments supplied for monitoring and protection of the packages are not accurate enough to achieve the small uncertainty margins necessary for a field test. Whenever possible, laboratory quality instrumentation should be installed for the tests. The accuracy of the instruments and the calibration procedure should be such that the measurement tolerances can be eliminated from future discussions regarding the performance of the unit.

One should note that test tolerances need to be clearly distinguished from building tolerances. Building tolerances cover the inevitable manufacturing tolerances and the uncertainties of the performance predictions. The actual machine that is installed on the test stand will differ in its actual performance from the predicted performance by the building tolerances. Building tolerances are entirely the responsibility of the manufacturer.

Test tolerances, on the other hand, are an expression of the uncertainty of the measuring and testing process. Namely, a machine tested with an 84 percent efficiency may have an actual efficiency somewhere between 82 percent and 86 percent.

Purpose of the Test

The customer and the manufacturer should agree upon and document the parameters of interest for the test, along with the criteria (minimums and maximums) for acceptance. Gas turbine power and gas compressor efficiency generally are the primary parameters, while compressor flow range, surge margins, and engine fuel flow are examples of other common performance parameters.

The method to calculate the parameters of interest from the measurements also needs to be defined in the test agenda, along with the applicable test specifications and the length of time expected to be allowed for measurements.

Turbomachinery performance tests usually are conducted independently of mechanical, vibration, emissions, and controls tests. However, if these tests are to be linked, the reasons should be clearly specified in the agenda.

The objective of the test is typically to verify acceptance criteria such as heat rate, specific fuel consumption, or the specific package fuel consumption,

$$SFC = \frac{W_f * LHV}{P} \qquad SFC_{packg} = \frac{W_f * LHV}{P * \eta^*} \tag{1}$$

the shaft power P of the engine and the gas power P_g of the compressor,

$$P_g = W \cdot H \tag{2}$$

the compressor efficiency η, compressor actual head H and flow W.

Typical conditions to be observed for these measurements are:

- Turbine
  - Ambient temperature: T_{amb}
  - Ambient pressure: p_{amb}
  - Power turbine speed: N_{PT}
  - Fuel flow: W_f
  - Fuel gas composition: -
  - Inlet and exhaust pressure loss: dp_i, dp_e
  - Turbine inlet temperature: T_1
- Compressor
  - Suction temperature: T_S
  - Suction pressure: p_S
  - Inlet flow: Q, W
  - Gas composition: -
  - Discharge pressure: p_d
  - Discharge temperature: T_d

Using equations of state and mixing rules, these parameters are also used to compute enthalpies h_s and h_d for suction and discharge conditions, and the enthalpy h_d^* for the isentropic discharge condition. The fuel gas composition allows the determination of the lower heating value E_f of the fuel and is also used to calculate the thermodynamic properties of the combustion gas.

In a very early stage of the project, discussions about necessary instrumentation and the site preparation to allow for the installation of the instrumentation (e.g., flow metering runs, thermowells, pressure taps) should be started. In this phase, the trade off between various options of installing instrumentation and the effect on the conduction of the test can be evaluated.

CORRECTION OF TEST CONDITIONS TO ACCEPTANCE CONDITIONS

Since the tested package will invariably be tested under conditions that deviate from the acceptance conditions, an adjustment must be made. This is accomplished by conducting the tests following the laws of similarity theory, i.e., the flow through the machines must be similar for the test and the acceptance condition. Preferably, the test conditions should follow the acceptance conditions as close as possible. It must be noted that the governing conditions for the laws of similarity are different for gas turbine and compressor. The correction, therefore, typically has to be performed for the compressor and the gas turbine independently.

Similarity Conditions for Compressors

For the compressor, similarity is accomplished if the following similarity parameters are the same for test and acceptance conditions:

Flow coefficient

$$\varphi = \frac{Q_s}{\frac{\pi}{4} D^2_{1,tip}\, u} \tag{3}$$

Head coefficient (isentropic or polytropic)

$$\psi^* = \frac{H^*}{\frac{u^2}{2}} \qquad \psi^p = \frac{H^p}{\frac{u^2}{2}} \tag{4}$$

Machine Mach number

$$Ma_u = \frac{\pi D_{tip} N}{\sqrt{k_s Z_s R T_s}} \tag{5}$$

Machine Reynolds number

$$Re_u = \frac{\pi D_{tip} N\, b_{tip}}{V_s} \tag{6}$$

Isentropic exponent

$$k = \left(\frac{v\,\delta p}{p\,\delta v}\right) \tag{7}$$

Ratio of volume flow ratios (Appendix A)

$$(Q_1/Q_2)_t = (Q_1/Q_2)_a \tag{8}$$

The comparison of the actual process with a polytropic process has, compared to the isentropic process, the advantage that the efficiency for an aerodynamically similar point is less dependent on the actual pressure ratio. However, it has the disadvantage that the polytropic head for a given set of operating conditions depends on the efficiency of the compressor, while the isentropic head does not.

Typically, not all similarity parameters can be brought in accordance with the desired acceptance criteria, especially when the gas composition during the test is different from the design gas. The most important parameters are head and flow coefficients, and the ratio of volume flow ratios. When keeping the flow coefficient the same as for the design case, the velocity triangles at the inlet into the first stage remain the same. Together with the head coefficient, this defines a singular operating point of the compressor, as long as the fan law remains applicable. If, in addition, the volume flow ratios between inlet and outlet are kept the same as for the design case, the velocity triangle at the outlet of the compressor will also be the same. Generally, this requirement involves keeping the same machine Mach number and keeping the same average isentropic exponents over the machine. The Reynolds number similarity is for many applications of lesser importance, since the Reynolds numbers for most applications are relatively high and clearly in the turbulent flow regime; also the loss generation in centrifugal compressors is only partially due to skin friction effects (i.e., effects that are primarily governed by Reynolds numbers). ASME PTC10 [1] allows the following deviations between design and test case for the parameters listed in Table 1.

Table 1. Acceptable Departures of the Test Conditions from Design Conditions.

Variable	Symbol	Departure
Inlet pressure	ps	5%
Inlet temperature	Ts	8%
Specific gravity of gas	SG	2%
Speed	N	2%
Capacity	Q_i	4%
Inlet gas density	ρ_i	8%

In general, as long as the deviations between test and design stay within these limits, a simple correction based on the fan law can be used. Namely, the test point must be at the same combination of φ and ψ (Equations (3) and (4)) as the design point. The limitations of the fan law are covered by Brown [7].

VDI 2045 [5] provides very specific guidelines about the deviations in volume ratio. If the volume ratio between acceptance criteria and test exceeds ± 1 percent, additional tolerances have to be applied. For this case, based on curves in VDI 2045 [5], the acceptable ratio of the relative speeds X, (or machine Mach numbers) between acceptance criteria and test:

$$X = \frac{\left(\frac{N}{\sqrt{RZ_s T_s}}\right)_t}{\left(\frac{N}{\sqrt{RZ_s T_s}}\right)_a} = \frac{Ma_{u,t}}{Ma_{u,a}} \tag{9}$$

can be determined.

If the test conditions are considerably different from the design conditions, (i.e., outside of the limits established in ASME PTC 10 [1] (Table 1)), easy corrections for Mach numbers and volume flow ratios are not available. Often, the design programs of the compressor manufacturer can be used to recalculate the compressor performance for the changed design conditions. Colby [8] states, that especially for compressors in applications where the compressibility factors change rapidly from suction through discharge, the deviations allowed for a PTC 10 Class I test might still be too high to simulate field conditions in a factory test.

Similarity Conditions for Gas Turbines

The similarity conditions for the gas turbine follow different considerations: the most influential characteristics are the inlet temperature, the power turbine speed, the ambient pressure, and, to a lesser degree, the fuel gas composition and the relative humidity.

> It is necessary to have the test conditions within limits agreed to by the parties to the test to avoid running the gas turbine at extreme conditions far from its design or rated condition, which would make the determination of accurate corrections impossible. . . The off-design characteristics of each gas turbine engine are unique. Hence, the manufacturer's . . . performance curves for the particular engine must be used to correct the actual test data to rated or standard conditions [3].

Since a gas turbine engine consists of three pieces of turbomachinery (compressor, gas generator turbine, and power turbine) plus a combustor, the application of similarity considerations as previously described for gas compressors, is virtually impossible. The goal would be to operate the engine in such a way that the component efficiencies are the same as for the acceptance point. For the engine compressor, this is achieved by maintaining identical corrected speeds, thus maintaining the same Mach numbers. For the gas generator turbine, this is similarly achieved by maintaining the same temperature ratio. The power turbine, since it is not mechanically connected to the gas generator shaft, will invariably run at a nonsimilar operating point. And, since the fuel gas composition during the test might differ from the design values, the gas behavior in the hot section can be different. Although a rough correction for ambient temperatures and ambient pressures is possible by:

$$T_{3t} / T_{3a} = T_{1t} / T_{1a}$$

$$N_{GPt} / N_{GPa} = \sqrt{\frac{(kRT_1)_a}{(kRT_1)_t}} \tag{10}$$

the only way of correcting the operation of gas turbines accurately is by using manufacturer software that models the engine, or manufacturer supplied correction curves. This is especially important if part load operating points were agreed upon. Since many modern engines are controlled by the gas generator speed, the firing temperature, and, possibly, variable guide vanes, simple curve matches are not sufficient to describe the engine performance variation due to different operating points.

In particular, it needs to be noted that the load of the engine must be similar under test conditions and acceptance conditions. This may lead to additional test points where the compressor is operated at other than the acceptance operating points. The reason is that the engine performance is very sensitive to the ambient conditions (pressure and temperature), while the performance of the compressor is not.

A deviation, for example, of 20°F from design ambient temperature has hardly any influence on the operating point, and especially the power consumption, of the gas compressor. For the engine, it may create the difference of operating at full load or at 95 percent load. Since the heat rate of the engine is sensitive to part load, the results can be quite different for both cases, especially for engines that bleed air at part load operations (e.g., for emissions control).

Based on the correction methods mentioned above, it will always be possible to come to test conditions that allow a meaningful demonstration of the acceptance criteria.

EXECUTING THE TEST

Steady State Conditions

Stable conditions are critical for a good test [9]. If the flow process is not in steady state, the conservation equations for flow and energy have to take the storage effects into consideration. Component temperatures and gaps between stationary and rotating parts show distinct time lags [10]. This cannot be resolved by using the "normal" test instrumentation. The test setup cannot account for time dependent effects. All test codes, therefore, allow only a certain fluctuation of measured parameters. No matter how small these fluctuations are, they will always increase the error margin for the test data. It is sometimes recommended to average the sampled data to eliminate these errors. As shown in the next example, since all the governing equations describe nonlinear relationships, the averaging of data will not resolve this problem.

Stable conditions are especially critical for temperature measurements. Temperature probes are often inserted into thermowells and reach equilibrium by convection. Convective heat transfer is not instantaneous. It is, therefore, necessary to maintain the same operating conditions for a longer period of time until the equilibrium is reached ("heat soak"). In addition, the large heat storing capacity of the compressor casing will need time to reach a new equilibrium after operating conditions change.

Steady state operations not only includes stability of pressures, temperatures, and speeds, but also the stability of the gas composition, alas of the molecular weight of the gas. The effect on measured φ–ψ curves during changing gas compositions is shown in Figure 3.

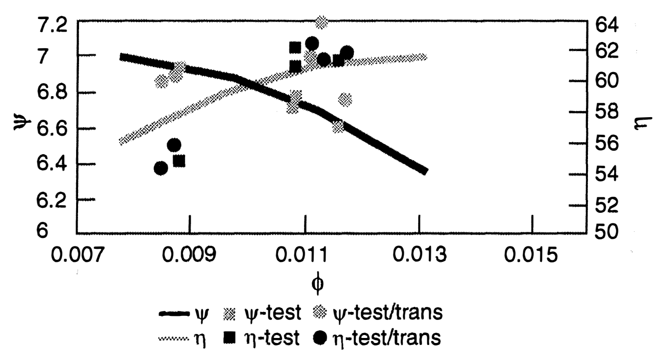

Figure 3. Effect of Transient Conditions on Compressor Test Results.

PTC 10 [1] allows certain fluctuations while taking data (Table 2). However, for pipeline applications, it is recommended to keep the fluctuations for inlet and discharge pressure below one percent.

Furthermore, care must be taken when averaging the data. To get correct results, the averaging should not be performed on the raw test data (e.g., pressures and temperatures), but only for the results (e.g., efficiency, power, etc.). Since many relationships are nonlinear, averaging the measured parameters and calculating the results with these averages will create different results than calculating the results for each sample point and averaging the results. Only the latter procedure will yield correct averages. Consider the following typical relationships:

Table 2. Allowable Fluctuation of Test Readings During a Test Run (From ASME PTC10 [1]).

Measurement	Fluctuation
Inlet pressure	2%*
Discharge pressure	2%*
Orifice differential pressure	2%
Orifice temperature	0.5%
Inlet temperature	0.5%
Speed	0.5%
Torque	1.0%
Specific gravity, test gas	0.25%

* Especially for low pressure ratio compressors, one percent should be used.

Orifice flow

$$Q = D\sqrt{\Delta p} \tag{11}$$

Efficiency

$$\eta = f\left[\left(\frac{p_d}{p_s}\right)^{\frac{k-1}{k}}, \Delta T\right] \tag{12}$$

Power

$$P = f\,(N_{GP}) \tag{13}$$

In these examples, averaging the values for Δp, pressure ratio, or engine speed N_{GP} is only permissible for steady state conditions. The expected engine power for 98 percent gas producer speed is not the arithmetic average between the power for 100 percent and 96 percent speed. Neither is the expected efficiency of a compressor for a pressure ratio of 2.0 the average of the efficiency for the pressure ratio of 1.9 and 2.1.

Datum averaging by itself does not remove the fundamental requirement of steady-state conditions.

Data Acquisition

Field testing has been simplified in recent years by the widespread application of portable computers that have introduced powerful analytical tools directly to remote locations where standard digital controls measure and display all necessary parameters.

In particular, it is now possible to monitor all measured parameters, such as pressures, temperatures, and speeds with a specially programmed laptop computer. A typical test setup is shown in Figure 4. It is also possible to extract some of the necessary data from the unit control system (UCS). An analog signal from a pressure transmitter or a thermocouple is transformed into digital data. The advantage of this procedure lies in the fact that all data for one measuring point can be taken at virtually the same instant, thus eliminating measurement inaccuracies due to unsteady operating condition fluctuation. This becomes especially important when correlating the gas turbine performance and the compressor performance. Modern portable computer data acquisition systems also allow a large number of data samples to be taken for each test point, thus reducing the overall statistical error. Setup time and especially test time get reduced dramatically. The data acquisition system in Figure 4 can be adapted to virtually all configurations of gas turbine driven compressor trains and is also easy to transport.

The practical requirements for temperature, flow, and pressure measurements are shown in Figure 5. The taps for dynamical pressure can be omitted, since the flow velocity can be determined by the volume flow measurement. The presented configuration, essentially taken from PTC 10 [1] differ somewhat from the VDI 2045 [5] recommendations. PTC 10 [1] recommends four pressure

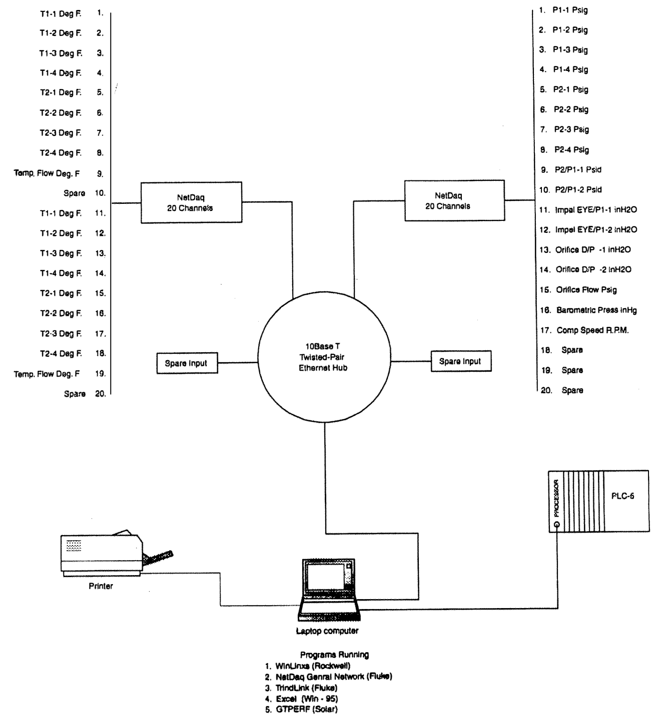

Figure 4. Typical Data Acquisition Setup for Automated Field Testing.

taps and four temperature stations for inlet and discharge side. The authors recommend the PTC10 [1] requirement, since in most applications the measurement stations are relatively close to elbows. Since the maximum error due to flow nonuniformities is in the order of the dynamic pressure, the decision has to be made based on the order of that pressure relative to the total pressure. The flow measurement devices, such as orifice plates, nozzles, and venturi nozzles, require runs of a certain straight length. The requirements are shown in Appendix D. Since a considerable distance may lie between the flow measurement device and the compressor inlet, the gas temperature and pressure need to be measured at the flow measuring device in addition to the measurement locations close to the compressor inlet.

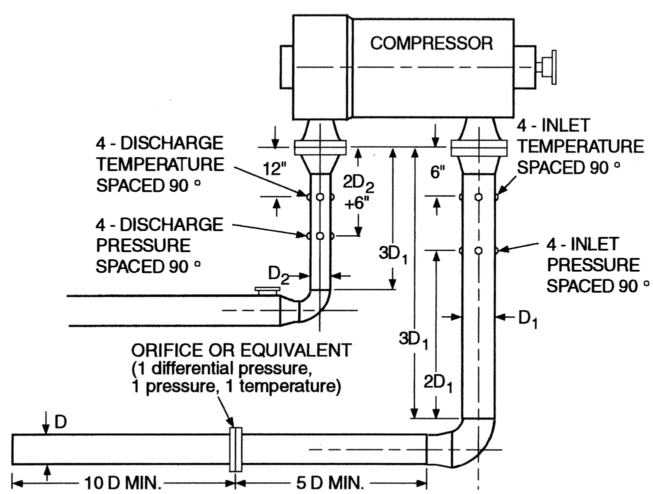

Figure 5. Practical Recommendations for Field Test Instrumentation.

DATA EVALUATION

Gas Properties

The equations for calculating head, efficiency, flow, and horsepower in a package require the knowledge of parameters (e.g., compressibility factor, specific heats, isentropic exponents, mole weight) that depend on the gas composition. These parameters can be calculated by using equations of state with mixing rules. Since different equations of state will yield different values for density, enthalpies, and entropies, the equation of state has to be agreed upon before the test. The effect is shown in Table 3 of different equations of state on the results for a given set of typical test data. Beinecke and Luedtke [11] have conducted thorough evaluations on the accuracy of the Lee-Kesler-Ploecker (LKP) method, the Benedict-Webb-Rubin-Starling (BWRS) method, and the Soave-Redlich-Kwong (SRK) method. All of them can predict the properties of hydrocarbon mixtures quite accurately over a wide range of pressures [11]. Still, deviations of 0.5 percent and more in the values for the compressibility factor Z are common. Even more important than the compressibility factor is the calculation of the enthalpy and entropy using the EOS. Since derivatives of the EOS have to be used to perform these calculations [12], the deviations can be even larger than for the compressibility factor (Table 3).

Table 3. Enthalpy Difference, Isentropic Enthalpy Difference, Efficiency, and Compressibility Factors Calculated with Different EOS.

EOS	H (kJ/kg)	H*(kJ/kg)	$\eta^*=H^*/H$	Z_1	Z_2
LKP	117.86	92.85	.788	.8954	.9207
SRK	117.93	93.58	.794	.8976	.9313
BWR	117.90	92.39	.784	.8928	.9168
PR	115.03	90.77	.791	.8748	.8998

Gas: 90% CH_4, 5% C_2H_6, 2% C_3H_8, 2% N_2, 1% CO_2, p_1 = 50 bara, p_2 = 100 bara, T_1 = 20°C, T_2 = 82°C.

For the same measured conditions shown in Table 3, four different equations of state deliver four different results. While all compressibility factors are relatively close together (except for Peng-Robinson), and even the isentropic efficiency differs by only one point, the highest and lowest enthalpy H are almost 2.5 percent apart. Since the enthalpy is used (together with the mass flow) to calculate the compressor power (Equation (2)), considerable differences can arise from the choice of equations of state. It should be noted that even the results for the same EOS may differ from program to program, because sometimes different mixing rules are used to calculate the constants in the EOS.

Comparison of Factory Test and Field Test

While testing a gas compressor in the field can yield similar measuring tolerances as the factory test, given that the field test is conducted using the same standards as for the factory test, the field testing of the engine will typically yield higher measuring tolerances than the measuring in the factory.

The main reason lies in the methodology of measuring the shaft power: in the factory, the shaft power is measured by running the engine against a dynamometer or a water brake. The power turbine applies torque onto the water brake or dynamometer. A load cell is used to measure the reaction force on the casing. The shaft power P is calculated by multiplying the measured torque and the measured shaft speed:

$$P = \tau\,(2\,\pi\,N) \tag{14}$$

In the field test, the shaft power is determined in one of the following ways:

- Using a torque measuring coupling between power turbine and driven equipment
- Using the calculated power of the driven equipment (heat balance)
- Concept of verification with a redundant measurement

While a torque metering coupling can achieve almost a similar accuracy as the factory test method, the power input into the driven equipment is subject to much higher measuring tolerances, as described herein.

The last method takes advantage of the conservation of energy in a thermodynamic system, thus requiring that the in flowing energy be balanced by the energy leaving the system:

$$w_1h_1+w_fE_f\eta_{comb.}+w_fh_f = w_7h_7+P+E_r+E_m \tag{15}$$

The mass flow and enthalpy of the air at the gas turbine inlet, the fuel flow, the fuel enthalpy, the lower heating value of the fuel, and mass flow and enthalpy of the exhaust gas can be measured. The radiated heat energy and the mechanical losses (that are leaving the system through the lube oil) can be estimated, and will be, in any case, rather small. The combustion efficiency can be estimated, as well, and should be about 98 percent. Therefore, the shaft power of the turbine P can be calculated. Essential for this method is the possibility to measure the airflow into the engine, which in the field is usually not possible with the necessary accuracy.

During the field test, the engine is operated at the prevailing ambient conditions. To correct the measured power and heat rate to acceptance conditions, a set of engine specific performance curves (Figures 6, 7, 8, and 9) is used, showing power and heat rate as a function of ambient temperature, with correction factors for barometric pressure, inlet and exhaust losses, and power turbine speed. The observed values are compared with predicted values for the same ambient conditions. Then, the percentage difference between observed and predicted values for the test conditions can be applied to the predicted values for the acceptance conditions. The same procedure, but with higher accuracy, is possible by using the computer programs that generate the curves.

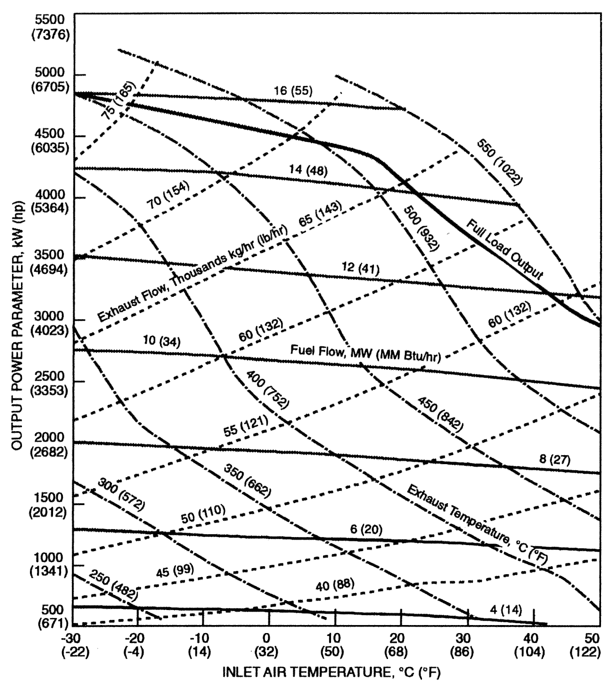

Figure 6. Typical Gas Turbine Performance Map for Power, Fuel Flow, Exhaust Flow, and Exhaust Temperature for Different Ambient Temperatures.

The uncertainties related to the measurement of the gas compressor required power will be discussed later. One problem

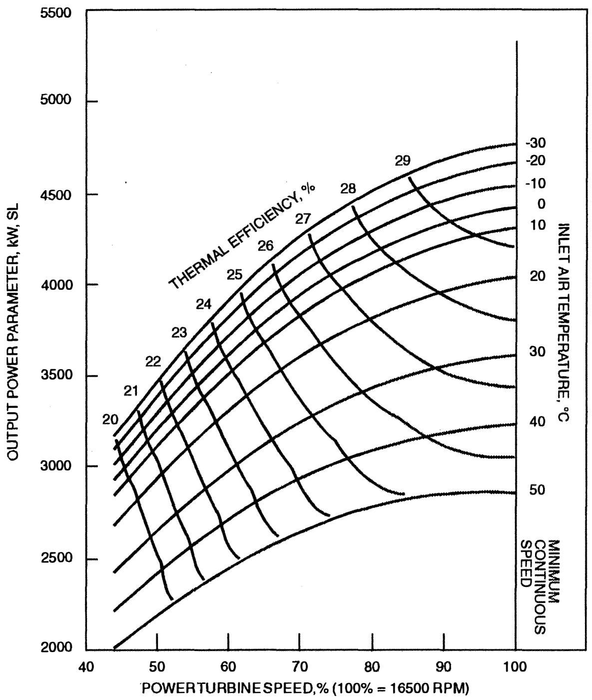

Figure 7. Typical Gas Turbine Performance Map for Full Load Power and Thermal Efficiency Vs *Power Turbine Speed and Ambient Temperature.*

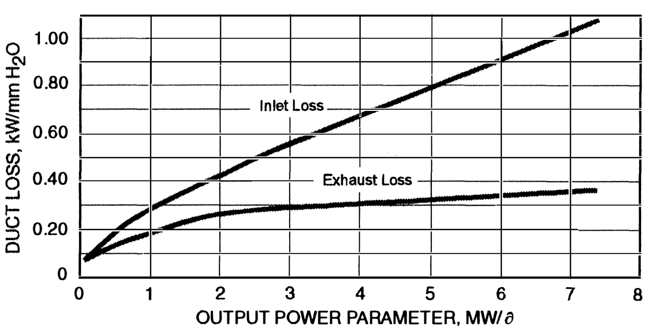

Figure 8. Correction Factor for Inlet and Exhaust System Losses.

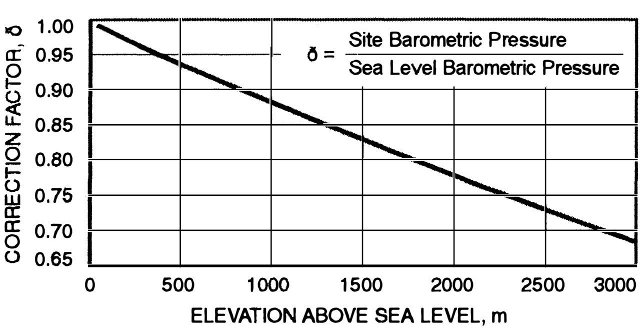

Figure 9. Correction Factor for Site Elevation.

that occurs particularly in low flow, high pressure applications will be discussed here: while compressing the gas from suction to discharge pressure, a certain amount of gas escapes through seals from the main flow and gets recirculated to the suction side. Many compressor manufacturers include this effect already into their performance predictions. Due to recirculation, the actual amount of gas that the compressor compresses is larger than the amount of gas flowing through the flow measuring device. The recirculated gas will also have a higher temperature than the incoming gas. The effect of recirculation is treated in Appendix C. It might lead to underestimations of the actually absorbed compressor power, which falsely reduces the engine efficiency. Especially for pipeline applications, the effect of recirculation can be neglected.

FIELD TESTING UNCERTAINTIES

For the uncertainty analysis, it is assumed that all measurement parameters can be considered to be independent and that parameters have associated statistical bounds (such as a 95 percent confidence interval Δu) rather than absolute limits of errors. All parameters are also assumed to have Gaussian normal distributions around their respective mean values such that the uncertainties can be properly combined using the root-square sum method [13]. However, an uncertainty correction is added for parameters that have sample sizes smaller than 30; i.e., the uncertainty is widened for individual parameters to account for a Student-t type distribution [14]. The total uncertainty ΔF, for a given function $F = f(u_1, u_2, ..u_n)$ is, thus, determined from [15]:

$$\Delta F = \sqrt{\left(\Delta u_1 \frac{\partial f}{\partial u_1}\right)^2 + \left(\Delta u_2 \frac{\partial f}{\partial u_2}\right)^2 + \ldots \left(\Delta u_n \frac{\partial f}{\partial u_n}\right)^2} \qquad (16)$$

For this method, the overall uncertainty ΔF has the same statistical meaning as the individual uncertainties Δu. Namely, if Δu represents a 95 percent confidence, then the result for the total uncertainty ΔF is also a 95 percent confidence interval. VDI 2045 [5] and VDI 2048 [6] propose to use a deadband approach to introduce the test uncertainties into the evaluation. This has the advantage of using a physically sound way of introducing the unavoidable uncertainty of the test results.

Note that since it is not possible to distinguish between bias errors and data scatter in a field test environment, they will not be treated independently.

Compressor Performance Equations

The parameters that are measured during a centrifugal compressor field test are:

- The compressor inlet and outlet stagnation temperature T_s and T_d
- The inlet and outlet static pressure p_s and p_d
- The actual inlet volume flowrate Q

Furthermore, knowledge of the gas composition is required to determine, based on a variety of available equations of state (SRK, LKP, BWRS, etc.), the gas molecular weight MW, the specific heat ratio as a function of static temperature γ(t), and the gas compressibility factor as a function of static pressure and temperature Z(p, t). Using this information, the following simplified equations can be used sequentially to determine head H, efficiency η, and required driver power P.

Gas Constant: $$R = \frac{R_{Universal}}{MW} \qquad (17)$$

Density: $$\rho = \frac{p}{R \cdot Z \cdot T} \qquad (18)$$

Specific Heat: $$c_p = Z \cdot R \cdot \left(\frac{\gamma}{\gamma - 1}\right) \qquad (19)$$

Head = Actual Enthalpy Difference: $H = c_p(T_d)\, T_d - c_p(T_s)\, T_s$ (20)

Isentropic Outlet Temperature: $T_d^* = T_s \cdot \left(\frac{p_d}{p_s}\right)^{\frac{\gamma-1}{\gamma}}$ (21)

Isentropic Head: $H^* = c_p(T_d^*)\, T_d^* - c_p(T_s)\, T_s$ (22)

Isentropic Efficiency: $\eta^* = \frac{H^*}{H}$ (23)

Driver Power: $P = \frac{H \cdot Q \cdot \rho}{\eta_M} = \frac{H \cdot Q \cdot p_s \cdot MW}{\eta_M \cdot R_{Universal} \cdot T_s}$ (24)

Note that for centrifugal compressor field tests one often assumes that stagnation and static values for pressure and temperature are very close, so that their difference can be neglected for the performance calculation. Care should be taken with this assumption, though; this is only a valid assumption as long as compressor local inlet and outlet local flow Mach numbers are below approximately 0.4. For example, in a typical pipeline compressor, the local outlet flow Mach number is approximately 0.3, which leads to a stagnation-static pressure difference of 0.05 percent and temperature difference of 0.01 percent. Based on the uncertainty equations that are derived below, one can show that these deviations lead to a negligible compressor isentropic efficiency uncertainty of 0.12 percent. On the other hand, if the Mach number approaches 0.6, typical pressure and temperature errors are 0.24 percent and 0.6 percent respectively, and the isentropic efficiency uncertainty can reach 1.2 percent. For the sake of brevity, it is assumed herein that stagnation and static values are identical.

In reality, the above simplified equations are not used in this form to determine compressor performance, but rather represent a close functional description of the actual physical equations. Nonetheless, since the total uncertainty is primarily based on the product of the individual uncertainty value and the gradient (df/du) of the governing equation, the simplified functional form is completely adequate to perform an accurate uncertainty analysis.

Compressor Uncertainty Equations

By substituting Equations (17), (18), and (19) into Equation (16) and letting $L = (\gamma - 1)/\gamma$, the following relation for the specific heat uncertainty Δc_p is obtained:

Specific Heat:

$$\Delta c_p = \sqrt{\left(\Delta Z \cdot \frac{R_{Universal}}{L \cdot MW}\right)^2 + \left(\Delta L \cdot \frac{R_{Universal} \cdot Z}{L^2 \cdot MW}\right)^2 + \left(\Delta MW \cdot \frac{R_{Universal} \cdot Z}{L \cdot MW^2}\right)^2} \quad (25)$$

The above Equation (25) is valid if the physical gas properties, specific heat ratio, compressibility factor, and molecular weight are directly determined from laboratory experiments. However, if these physical properties are calculated from a given gas composition via an equation of state, they cannot be considered to be independent.

For this case, Equation (25) has to be slightly modified; namely, the absolute value sum of the individual uncertainty terms instead of the root-square sum should be taken to account for their functional dependency:

$$\Delta F = \left|\Delta u_1 \frac{\partial f}{\partial u_1}\right| + \left|\Delta u_2 \frac{\partial f}{\partial u_2}\right| + \ldots \left|\Delta u_n \frac{\partial f}{\partial u_n}\right| \quad (26)$$

This leads to a more conservative estimate of the uncertainty [13]. Furthermore, a physical property uncertainty, due to the effect of applying uncertainties in T and p to the nonideal gas state equation, has to be included; i.e., since there is a measurement error in T and p, there will be an added error in determining c_p from the gas equation. This uncertainty is most conveniently obtained numerically by varying temperatures and pressures parametrically in the gas equation and, thus, determining the gradients $d\gamma/dT$, $d\gamma/dp$, dZ/dT, and dZ/dp indirectly. Recognizing that $d\gamma/dT = dL/dT$ and $d\gamma/dp = dL/dp$, one can easily determine corrections for ΔZ and ΔL:

$$\Delta L = \sqrt{\left(\Delta T \cdot \frac{\partial \gamma}{\partial T}\right)^2 + \left(\Delta p \cdot \frac{\partial \gamma}{\partial p}\right)^2} \qquad \Delta Z = \sqrt{\left(\Delta T \cdot \frac{\partial Z}{\partial T}\right)^2 + \left(\Delta p \cdot \frac{\partial Z}{\partial p}\right)^2} \quad (27)$$

The uncertainty in c_p is also affected by the variation of the gas properties during the duration of the test. This effect is again mathematically difficult to describe but can be easily handled numerically using a procedure similar to the one shown above for the variations in T and p. It is beyond the scope herein to list all possible gas composition variations; however, it is important to realize that they can strongly affect Z, L, and MW. For example, if a simple gas mixture of 90 percent methane and 10 percent ethane has a constituency uncertainty of one percent, then the resulting uncertainties in ΔMW, ΔZ, and ΔL are 0.9 percent, 0.05 percent, and 0.07 percent, respectively.

Finally, a comment should be made regarding the consistent application of the gas state equations. For identical gas compositions, one finds significant physical property output differences between the commonly employed gas equations of state (SRK, BWRS, and LKP). Beinecke and Luedtke [11] noted typical differences in Z values of 0.5 percent to 1.5 percent when comparing the output from SRK, BWRS, and LKP for a standard pipeline application. Thus, if one compressor manufacturer employs the SRK equation and another the BWRS gas equation, predictions for identical compressors may vary significantly. Also, if the compressor measurements are being employed to guarantee the driver (gas turbine) output performance, deviations of the gas equation output from actual physical values will lead to incorrect results for the required power. Errors in gas characteristics due to the equation of state do not have to be included in the calculation of measurement tolerances for the compressor efficiency, since, and as long as, they are used in the design of the machine. Therefore, the errors in these equations will eliminate themselves. If both for the test and the design of the compressor the same equation of state with the same mixing rules is used, then no allowance for errors in gas characteristics needs to be made. If the test is conducted using a different equation of state than was used for the design of the compressor, additional tolerances need to be made. Also, if the compressor power measurement is used to determine the engine shaft power, the uncertainties due to the equation of state have to be considered.

The dependency of the gas characteristics on measured (and thus being subject to errors) parameters needs to be taken into consideration. For example: while the error in the compressibility factor Z, due to the equation of state, will be eliminated, the error due to erroneous values of pressures and temperatures has to be considered.

By substituting Equation (25) into (20) the uncertainty of the compressor head is determined:

Actual Head:

$$H = \sqrt{(\Delta c_{pT_d} \cdot T_d)^2 + (\Delta T_d \cdot c_{pT_d})^2 + (\Delta c_{pT_s} \cdot T_s)^2 + (\Delta T_s \cdot c_{pT_s})^2} \quad (28)$$

Uncertainties in the temperature T originate from the five following major sources of error:

- Location: incorrect position of the thermal sensor in the gas stream
- Installation: wall conduction heat transfer to and from the sensor due to inadequate insulation
- Calibration: instrument drift, nonlinearities, cold junction, and reference temperature errors

- Device: inherent accuracy limitations of the sensor device
- Acquisition: amplifier, transmission, noise, read, and analog-digital conversion errors.

The first three of these errors can be minimized easily in production or laboratory test facilities. However, for field testing this is more difficult; namely, time and cost constraints can force the test engineer to accept field test arrangements with improperly located, installed, and calibrated instruments. For example, a minimum of four circumferentially spaced pressure tabs should be employed to accurately measure gas pressures behind a pipe elbow. Due to the above mentioned constraints, typically, only one tap is employed in field testing. While it is often impossible to correct these problems during the short field test duration, it is still important to recognize them and to advise the customer of consequential measurement accuracy limitations. The ASME PTC 10 [1] code specifies proper installation and location of the temperature sensors and, thus, should be used as a guideline and reference when defining field test procedures. Typical values for the above five main sources of temperature measurement errors encountered during field tests are displayed in Table 4. All units are in degrees Celsius.

Table 4. Typical Magnitudes of Temperature Measurement Errors (°C).

Sensor	Location*	Installation*	Calibration	Device	Acquisition
Hg thermometer	0.1	0.05	0.0	0.03	0.1
Thermistor	0.1	0.1	0.2	0.02	0.05
Thermocouple	0.1	0.03	0.3	0.01	0.05
RTD	0.1	0.1	0.2	0.01	0.05
Infrared sensor	0.3	0.0	0.3	0.05	0.05

(* Assuming average of four devices equally spaced on circumference.)

Aust [16] achieved, under laboratory conditions, ± 0.2 K with shielded NiCrNi-thermocouple probes. He mentioned deviations that depended on the flow velocity. Since the flow velocities in the nozzle of gas compressors are clearly subsonic, inaccuracies due to recovery factors should be negligible. The high accuracy will not easily be achieved in the field, since Aust used compensation elements kept in an oil bath of 0°C ± 0.1°C. This procedure is not practical in the field. In any case, he found the calibration curves given in DIN43710 to describe the behavior of the thermocouples precisely. In the field, a total inaccuracy of ± 0.5 K seems achievable. Cleveland [17] shows instrument accuracies for thermometers to be 0.25 to 1.0 K, thermocouples 0.25 to 1.0 K and RTDs 0.0025 to 2.5 K. Schmitt and Thomas [16] report 0.5 K for RTDs. VDI 2045 [5] assumes a tolerance of ± 1.0 K for thermocouples and RTDs.

To obtain the total temperature uncertainty ΔT, the individual uncertainties have to be added using the root-square sum method. In Table 4, it is shown that the location, installation, and calibration errors are the dominant factors, while the device and acquisition errors are a smaller contribution to the total temperature error. Also, note that field test device and acquisition errors are significantly larger than values quoted by instrument manufacturers (>0.005 percent full scale). Again, the circumstances and limitations encountered in field test do not allow for ideal handling of the sensitive measurement instruments.

When substituting Equation (21) into Equation (16), the uncertainty for the isentropic (ideal) compressor outlet temperature is obtained:

Isentropic Temperature:

$$\Delta T_d^* = \sqrt{\left(\Delta T_s \cdot \left(\frac{p_d}{p_s}\right)^L\right)^2 + \left(\Delta p_d \cdot \frac{L T_s p_d^{L-1}}{p_s^L}\right)^2 + \left(\Delta p_s \cdot \frac{L T_s p_d^L}{p_s^{L+1}}\right)^2} \quad (29)$$

Uncertainties in pressure can also be categorized into the following five major groups of errors:

1. Location: incorrect position or alignment of the pressure probe in the gas stream
2. Installation: piping vibration transmission to the pressure pickup due to inadequate damping
3. Calibration: instrument drift, nonlinearities, hysteresis, and reference pressure errors
4. Device: inherent accuracy limitations of the sensor device
5. Acquisition: amplifier, transmission, noise, and analog-digital conversion errors

Again, error sources 1, 2, and 3 are significantly larger in field testing than in production or laboratory tests. Although the ASME PTC 10 code [1] provides clear guidelines for correct installation and location of pressure probes, one often finds these to be the main sources of pressure measurement errors. Some typical values for the five sources of pressure measurement errors encountered during field tests are presented in Table 5. All values are percent full scale.

Table 5. Typical Magnitudes of Pressure Measurement Errors (Percent Full Scale).

Sensor	Location*	Installation	Calibration	Device**	Acquisition
Simple static	2.0	0.05	0.3	0.1	0.01
2 Static	*1.0*	*0.05*	*0.3*	*0.1*	*0.01*
4 Static	*0.5*	*0.05*	*0.3*	*0.1*	*0.01*
Pitot	1.0	0.1	0.3	0.1	0.01
Kiel	0.2	0.1	0.3	0.1	0.01

*This error will depend largely on the uniformity of the flow at the measuring location.
**Also reported in Schmitt and Thomas [18] and by transmitter manufacturers.

Aust [16] achieved 0.1 percent of maximum value for a calibrated system for wall static pressures, not including location. The location should not cause an error larger than one percent of dynamic pressure. Boelcs and Suter [19] show the dependency of the wall static error on wall shear stress. Wall taps need to be exactly perpendicular and flush to the surface. No burrs or slag are acceptable. The authors recently tested a machine where the pressure taps were blown into the pipe with a cutting torch. The resulting spread of the four pressure transducers was almost 2.0 percent of the static pressure. Preferably, the wall tap should be followed by a larger diameter hole [19]. The ratio of dynamic to static pressure for most applications will be below one percent, since the pipe diameters (and the compressor nozzle diameters) are selected to avoid high flow velocities. Also, the distortion that is seen by the impeller will be reduced, due to the fact that the flow normally gets accelerated considerably between the nozzle and the impeller eye.

VDI 2045 [5] assumes 0.2 percent of full scale for transducers and gauges. It should be emphasized that the instruments should be selected in a way that the measured values are normally in the upper 25 percent of full scale. Liquid columns can be more precise, but the precision depends largely on the accuracy of reading the scales. One should take into account that field tests tend to last over many hours, thus any system that avoids the human factor in reading instruments should be preferred.

The uncertainty of the compressor efficiency $\Delta\eta^*$, is determined by substituting Equations (22) and (23) into Equation (16):

Isentropic Head:

$$H^* = \sqrt{(\Delta c_{pT_d^*} \cdot T_d^*)^2 + (\Delta T_d^* \cdot c_{pT_d^*})^2 + (\Delta c_{pT_s} \cdot T_s)^2 + (\Delta T_s \cdot c_{pT_s})^2} \quad (30)$$

Isentropic Efficiency: $$\Delta\eta^* = \sqrt{\left(\frac{\Delta H^*}{H}\right)^2 + \left(\Delta H \cdot \frac{H^*}{H^2}\right)^2} \quad (31)$$

Finally, the driver power uncertainty ΔP, is obtained by combining equations (24) and (16) to get the following:

Mass Flow: $$\Delta W^2 = \left(\Delta p_s \cdot \frac{MW \cdot Q}{R_{Universal} Z T_s}\right)^2 + \left(\Delta MW \cdot \frac{p_s Q}{R_{Universal} Z T_s}\right)^2$$
$$+ \left(\Delta Q \cdot \frac{p_s \cdot MW}{R_{Universal} Z T_s}\right)^2 + \left(\Delta Z \cdot \frac{p_s \cdot MW \cdot Q}{R_{Universal} Z^2 T_s}\right)^2 + \left(\Delta T_s \cdot \frac{p_s \cdot MW \cdot Q}{R_{Universal} Z T_s^2}\right)^2 \quad (32)$$

Power: $$\Delta P = \sqrt{\left(\Delta H \cdot \frac{W}{\eta_M}\right)^2 + \left(\Delta W \cdot \frac{H}{\eta_M}\right)^2 + \left(\Delta \eta_m \cdot \frac{H \cdot W}{\eta^2_M}\right)^2} \quad (33)$$

The flowrate uncertainty ΔQ, depends strongly on the device type employed for the measurements. A detailed discussion of flow measurement uncertainty is provided in ASME PTC 19.1 [2] and is, thus, not further discussed herein. In Table 6, some typical values of ΔQ from field testing experience are provided.

Table 6. Typical Magnitudes of Volume Flowrate Measurement Errors (Percent Full Scale).

	Measurement Error	Sensitivity to Location	Pressure Loss
Orifice	1.5% (0.5%)	medium to low	high
Venturi	1.5% (0.5%)	medium	moderate
Ultrasound	0.5%	medium to high	low
Vortex [17]	1%		
Turbine Flow Meter	1.0-2.0% (0.5%)	medium	high
Annubar	1.0-1.5% (0.5%)	medium to low	moderate

In parentheses are the values that can be obtained with lab quality calibrated devices in perfect arrangements.

ISO2314 [3] assumes 0.5 percent accuracy for orifice flow measurements. This seems to be unrealistic for field tests. Schmitt and Thomas [18] report an uncertainty for an orifice metering run per ISO5167 [20] of 1.4 percent.

All above values do not account for errors in density. The gas analysis has a certain error margin. Density errors of 10 percent are reported. Even for pipeline applications, error margins are one percent or larger [21]. Especially cumbersome is the analysis if the gas samples are taken far away from the compressor or upstream of a separator or knockout vessel [22].

A properly selected and calibrated fuel flow measuring device can be suitable to achieve measurement accuracies of ± 0.5 percent of the measured quantity [3]. However, the additional effort to get from one percent accuracy to 0.5 percent accuracy might not always be justified when the driven compressor is used for the power measurement. With, say, three percent accuracy of the power measurement, the engine heat rate will have a tolerance of 3.16 percent and 3.04 percent respectively, i.e., the relative gain is only 1/10 percent in accuracy. According to a recent publication about the efforts of one research institute to evaluate gas flow measuring methods [23], the measurement errors for orifices are one percent (mainly depending on piping configuration and diameter ratio of the orifice) and the measurement errors for turbine flowmeters are more than one percent (especially due to flow pulsations). This publication emphasizes the wide flow range and the very low pressure losses of ultrasonic flowmeters. Ongoing research is trying to quantify the sensibility of this device to distortions of the flow profile.

While errors in the measurement of the flow through a compressor will not influence the accuracy of the compressor efficiency (they will however, cause wrong information about the actual operating point of the machine), they play a significant role in determining the shaft power, as Fozi [9] points out. Inaccurate flow measurements can be identified by plotting the head-flow test data for an entire speed line on the predicted speed line. If the shape of the curves is the same but seems to be shifted to higher or lower flows, the flow measurement might be flawed. Flow measurement problems frequently occur, when multiple-compressor stations use only one accurate flow device for the whole station, which naturally will operate at the low end of its range when only one machine is operated for test purposes.

If a torquemeter is used, the total uncertainty for the engine power can be reduced to one percent. Even lower values are reported [18]. A torquemeter measurement also gives a good baseline for crosschecking the compressor performance.

By evaluating Equations (25) through (33), estimates of the total measurement uncertainties for the compressor efficiency, head, and required driver power can be obtained. However, one source of measurement uncertainty that is often overlooked is the uncertainty due to a finite sample size. The above uncertainty statistics, particularly Equation (16), is valid only for mean parameters with an assumed Gaussian normal distribution. This is a good assumption for measurements where sample sizes are larger than 30. But, for field tests, it is sometimes difficult to maintain a steady-state system operating condition for a time period adequate to collect 30 or more samples. For small sample sizes, it is more appropriate to assume a Student-t distribution instead of a Gaussian normal. Thus, an additional uncertainty due to a finite number of samples should be introduced into the total performance parameter uncertainty. This uncertainty is quantified by conservatively assuming that all individual uncertainty bands are within their 95 percent confidence intervals and then determining the percent difference between Gaussian and Student-t distribution limits. Some typical values of the added uncertainty due to limited number of samples for the compressor head, efficiency, and power are shown in Table 7. Note that the values in Table 4 are dependent on inlet/outlet temperatures, inlet/outlet pressures, and gas composition. The limited sample size uncertainty should be added to the total parameter uncertainty, using the root-square sum method.

Table 7. Parameter Uncertainty Due to Limited Sample Size (Percent).

Number of Samples	Head	Efficiency	Power
20	0.043	0.058	0.082
22	0.038	0.052	0.072
24	0.033	0.046	0.062
26	0.028	0.040	0.052
28	0.023	0.034	0.042
30	0.018	0.028	0.032

Turbocompressor Package Uncertainty

To complete the above field test measurement uncertainty evaluation, one also needs to look at the complete turbocompressor train (gas turbine and compressor efficiency) performance. On direct drive applications, the gas turbine shaft output power has to equal the compressor required power (P_{GT} = P). Thus, the following two equations can be used to define the gas turbine efficiency η_{TH}, and the total package uncertainty η_P:

$$\eta_{TH} = \frac{P}{W_{fuel} \cdot q} \quad (34)$$

$$\eta_P = \eta_I \cdot \eta_M \cdot \eta_{TH} \quad (35)$$

Where W_{fuel} is the fuel flow into the engine and q is the fuel heating value. The fuel flow is typically measured using an orifice plate in a metering run and the heating value is determined from the chemical composition of the fuel (often the centrifugal compressor discharge gas is used as the fuel gas). Based on the above equations, the corresponding gas turbine uncertainty $\Delta\eta_{TH}$, and package uncertainty $\Delta\eta_P$, are given by:

$$\Delta\eta_{TH} = \sqrt{\left(\Delta P \frac{1}{W_{fuel}q}\right)^2 + \left(\Delta W_{fuel} \frac{P}{W^2_{fuel}q}\right)^2 + \left(\Delta q \frac{P}{W_{fuel}q^2}\right)^2} \quad (36)$$

$$\Delta\eta_p = \sqrt{(\Delta\eta_I\eta_M\eta_{TH})^2 + (\Delta\eta_M\eta_I\eta_{TH})^2 + (\Delta\eta_{TH}\eta_M\eta_I)^2} \quad (37)$$

To complete the above Equations (36) and (37), the only additional information needed is the fuel flow uncertainty and the fuel heating value uncertainty. Since the fuel flow is measured in the same way as the flow through the gas compressor, uncertainty values in Table 6 can be used. Also, since the heating value is obtained directly from gas composition, the same percent uncertainty as was obtained for the specific heat (Equation (25)) can be approximately used, namely:

$$\frac{\Delta_q}{q} = \frac{\Delta c_p}{c_p} \quad (38)$$

One more factor needs to be considered: The uncertainty for calculating the package heat rate might be lower than for the shaft power, because T_{disch} is not required (direct calculation). But if the package heat rate needs to be corrected from test conditions to acceptance conditions, the shaft power is needed, because the correction for the compressor and for the engine has to be done separately (indirect calculation). The order of magnitude is established in the example in Appendix B. Due to very different ambient conditions (design 35°C, test 5°C), the engine is at 78 percent load during the test, although the design point was a full load point.

It shows that, except under conditions even more extreme than in the example, the difference between the direct and indirect calculation is negligible. To calculate the uncertainties independently, as in the last line, will lead to exaggerated uncertainties.

Uncertainty Calculation Examples and Comparisons to Field Test Results

Two examples are shown below to demonstrate how the total measurement uncertainty is obtained for some actual field test experiments. Also, a comparison to some factory test results is presented in Example B.

Example A

During field testing of a turbocompressor installed on a platform in the Atlantic, the following conditions were encountered:

- Compressor application: gas reinjection
- Instrumentation: single RTD and single pressure tab on each suction and discharge; orifice flowmeter for compressor flow and turbine flowmeter for engine fuel flow
- Test conditions: p_1 = 65 bar, p_2 = 226 bar, T_1 = 303° K, T_2 = 448.5° K (η^* = 62.0%), Q = 18.4 m^3/s
- Gas composition: fluctuating by five percent in constituency composition; during test 93.1 percent methane, 5.8 percent ethane, 0.4 percent propane, 0.15 percent I-butane, 0.25 percent n-butane, 0.3 percent nitrogen

The compressor field test uncertainty is predicted by evaluating Equations (25) through (33) with assumed measurement uncertainties as suggested in Tables 4 through 7 ($\Delta T_{1,2}$ = 0.25 K, $\Delta p_{1,2}$ = 2.31 percent, ΔQ = 1.0 percent, $\Delta\eta_M$ = 0.1 percent). The influence of the gas constituency fluctuation on the total uncertainty is accounted for by using the method as outlined in Equation (27), which yields ΔZ = 0.57 percent, $\Delta\gamma$ = 0.72 percent, and ΔMW = 0.08 percent for this case. From the resulting absolute uncertainty values for the specific heat, the actual head, the isentropic efficiency, and the required power (Δc_p, ΔH_A, $\Delta\eta_I$, $\Delta \dot{W}$) the percent total field test uncertainties are determined; namely: $\Delta c_p/c_p$ = 0.65 percent, $\Delta H/H$ = 2.69 percent, $\Delta\eta^*$ = 4.33 percent, $\Delta P/P$ = 3.67 percent.

Thus, total efficiency uncertainties of above four percent are seen for this set of field test measurements. The main cause for these very large uncertainties is the inadequate number of pressure tabs installed at the compressor suction and discharge. Increasing the number of pressure tabs from one to four on each suction and discharge would reduce the predicted compressor and power uncertainties to $\Delta\eta^*/\eta^*$ = 1.84 percent, $\Delta P/P$ = 1.95 percent, respectively; a significant measurement improvement.

To verify the applicability of the above uncertainty method, results from the Equations (25) through (33) are compared to some actual observed field test measurement deviations. The field test deviation is determined from the difference between the field test and the factory closed loop compressor test. Clearly, this deviation does not take into account all possible measurement errors that can occur in both the factory and the field tests, but simply represents a comparison of a higher (factory test) *vs* a lower (field test) accuracy measurement. Thus, one would not expect this deviation to be as large as the total calculated field test measurement uncertainty, which represents essentially a sum of all possible measurement errors.

Example B

During a turbocompressor field test in Germany, the following compression conditions were encountered:

- Compressor application: pipeline
- Instrumentation: multiple thermocouples and three pressure tabs on each suction and discharge; orifice flowmeters for compressor flow and engine fuel flow
- Test conditions: p_1 = 53 bar, p_2 = 80 bar, T_1 = 288°K, T_2 = 318°K (η^* = 86.0 percent), Q = 92.1 m^3/s, W_{fuel} = 0.257 kg/s, η_{TH} = 29.1 percent, η_P = 25.0 percent
- Gas composition: 91 percent methane, 4.9 percent ethane, 1.9 percent propane, 0.3 percent I-butane, 0.4 percent n-butane, 0.7 percent carbon dioxide, 0.8 percent nitrogen

After reducing the field test measurement data, the compressor efficiency η^* and power results P are seen to deviate by 1.36 percent and 1.10 percent from test cell measurements, respectively. As a comparison, when applying the above test conditions to Equations (25) through (35) and assuming measurement uncertainties as shown in Tables 4 through 7 ($\Delta T_{1,2}$ = 0.23 K, $\Delta p_{1,2}$ = 0.76 percent, ΔQ = 1.0 percent, $\Delta\eta_M$ = 0.1 percent), the following percent uncertainties are obtained: $\Delta c_p/c_p$ = 0.53 percent, $\Delta H/H$ = 1.10 percent, $\Delta\eta^*$ = 2.15 percent, $\Delta P/P$ = 1.81 percent. Similarly, evaluating Equations (21) and (22) for the package and gas turbine total uncertainties with ΔQ_{fuel} = 0.5 percent and Δq = 0.53 percent, yield $\Delta\eta_{TH}$ = 2.77 percent and $\Delta\eta^*_P$ = 2.39 percent, respectively.

Thus, the isentropic efficiency deviation is conservatively over predicted by 0.79 percent and the power deviation by 0.71 percent. Again, one must remember that the theoretical uncertainties as determined from Equations (25) through (33) yield results for the total possible performance parameter uncertainty range (which is a very conservative estimate), while the deviation is just a comparison between two different measurements. One would, thus, anticipate the significantly larger theoretical uncertainty values.

Another example is found in the literature [24]. A multistage gas compressor driven by a gas turbine was tested, using both the energy balance for the compressor and a torquemeter. At maximum speed, differences between the power derived from both methods were almost four percent.

Overall, the uncertainty equations are seen to predict measurement fluctuations well. However, the general experience with this method is that the uncertainty method conservatively over predicts errors by a small margin.

Parametric Studies and Trends

A number of parametric studies were undertaken to determine the effect of varying individual measurement variables on the total performance uncertainties. These studies also help to identify parameters that have the most significant effect on the total uncertainty. Hence, the effect of varying a number of field test measurement parameters on the efficiency and power uncertainty are evaluated. The compressor operating conditions as presented as Example A above were used as a basis for the studies below.

Results of varying the compressor head and suction temperature (keeping all other conditions, including efficiency, constant) on the efficiency and power uncertainty are shown in Figures 10 and 11. As the head is increased, the efficiency and power uncertainties decrease. The uncertainty rate of decrease is seen to be a direct function of the increasing compressor head magnitude. Also, the uncertainties are seen to be lower for increased compressor suction temperatures. Both of these results are expected and can be explained as follows: the larger the pressure or temperature suction-discharge differential, the smaller the relative influence of the individual measurement errors on the total performance uncertainties.

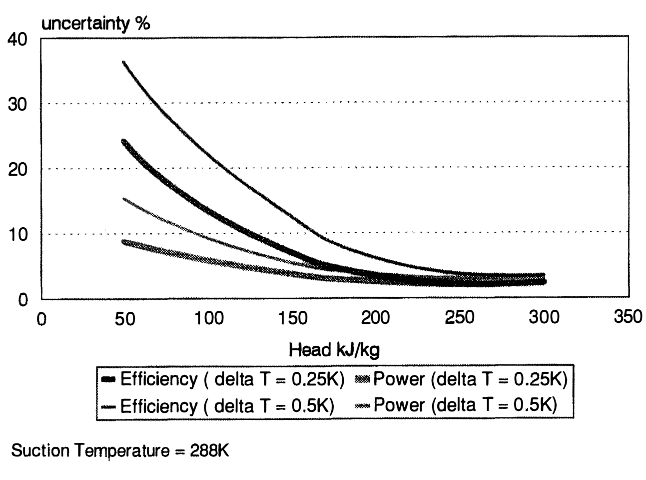

Figure 10. Efficiency and Power Uncertainty for Different Isentropic Head (Based on Example A), Suction Temperature 288 K.

Results in Figures 10 and 11 are interesting, but do not provide any guideline as to which uncertainty most significantly affects the overall performance parameter uncertainties. Thus, a quantitative comparison of the individual measurement uncertainties affect on the compressor isentropic efficiency uncertainty is shown in Tables 8 and 9. For this study, only one measurement uncertainty parameter (e.g., inlet temperature ΔT_S) was varied, while all others were left at the values described in Example A above.

Similarly, a quantitative comparison of the individual measurement uncertainties affect on the compressor required power uncertainty is shown in the numbers in Tables 10 and 11. Again, for this study only one measurement uncertainty parameter was varied, while all others were left at the values described in Example A above.

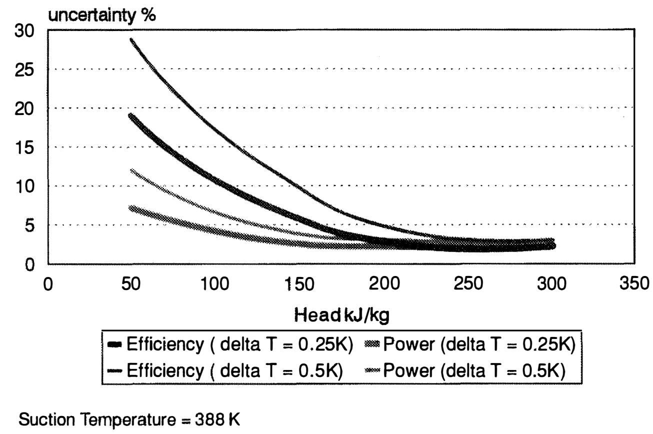

Figure 11. Efficiency and Power Uncertainty for Different Isentropic Head (Based on Example A), Suction Temperature 388 K.

Table 8. Compressor Isentropic Efficiency Uncertainty (Percent).

	Δp_s	Δp_d	Δ (Gas Composition)
0.5% Error	4.6	4.3	3.6
1.0% Error	4.7	4.7	4.7
1.5% Error	4.8	4.9	6.3
2.0% Error	4.9	5.2	7.9

Table 9. Compressor Isentropic Efficiency Uncertainty (Percent).

	ΔT_s	ΔT_d
0.1 K	3.3	3.5
0.2 K	4.0	4.0
0.3 K	4.7	4.4
0.4 K	5.4	4.8
0.5 K	6.0	5.1

Table 10. Compressor Required Power Uncertainties (Percent).

	Δp_s	Δp_d	Δ (Gas Composition)	ΔQ
0.5%	2.4	2.3	1.9	2.0
1.0%	2.5	2.5	2.5	2.5
1.5%	2.5	2.6	3.6	3.0
2.0%	2.6	2.7	4.2	3.5

Table 11. Compressor Required Power Uncertainties (Percent).

	ΔT_s	ΔT_d
0.1 K	1.7	2.1
0.2 K	2.2	2.4
0.3 K	2.5	2.6
0.4 K	2.7	2.8
0.5 K	2.8	2.8

Results are shown in Tables 8 through 11 that the temperature and gas composition uncertainty have the most significant effect on the overall performance uncertainties. Pressure measurements are seen to be less significant. However, due to the added complexity of pressure measurements, the field test pressure measurement uncertainties are often much larger than temperature measurement uncertainties and, thus, pressure measurement uncertainties should not be simply discounted.

CONCLUSIONS

The critical factors in determining correct results are:

- Steady state
- Pressure ratio
- Temperature
- Gas composition

The most critical success factor, however, is that a cognizant agreement between the parties is achieved prior to the test on how to conduct and evaluate the test. Even though the final test uncertainties can only be calculated after the test, agreement should be reached on the method of calculating them and a preliminary determination of these uncertainties should be conducted. Well before the test, an analysis should be performed that identifies the sources of measurement errors and aims for improvement of only those instruments that have a significant impact on the overall uncertainty.

The goal herein was to present some considerations on what can be expected from a field test and what the parties need to discuss in preparation of such a test.

NOMENCLATURE

b = Impeller tip width
β = Diameter ratio (for orifices and nozzles)
γ = Ratio of specific heats
D = Diameter
E = Energy
E_f = Lower heating value of the fuel
h = Enthalpy
k = Isentropic exponent
Ma = Mach number
MW = Molecular weight
N = Speed
n = Polytropic exponent
p = Pressure
P = Power
Q = Volumetric flow
q = Fuel heating value
R = Gas constant
Re = Reynolds number
ρ = Density
T = Temperature
τ = Torque
u = Velocity
υ = Kinematic viscosity
v = Specific volume
W = Mass flow
Z = Compressibility factor
η = Efficiency
H = Head
EOS = Equation of state
BWR = Benedict-Webb-Rubin
LKP = Lee-Kesler-Ploecker
PR = Peng-Robinson
RK = Redlich-Kwong
SRK = Soave-Redlich-Kwong
Z = Compressiblity factor

Subscripts:

a = Acceptance criteria
amb = Ambient
d = Discharge
f = Fuel
M = Mechanical
r = Recirculation
s = Suction
t = Test
TH = Thermal
tip = Impeller tip
PT = Power turbine
GP = Gas producer
1-7 = Refers to positions in the gas turbine
1 = First stage of the compressor

Superscripts:

* = Isentropic
p = Polytropic

APPENDIX A

The volume ratio:

$$\frac{Q_1}{Q_2}=\frac{\rho_2}{\rho_1}=\frac{p_2}{p_1}\frac{T_1}{T_2}\frac{Z_1}{Z_2} \tag{A-1}$$

means, that Q_1/Q_2 depends on a significant Mach number, the efficiency (which, by definition, will stay constant if similarity is achieved), the isentropic exponent, and the gas behavior. For an isentropic process, this becomes:

$$\frac{Q_1}{Q_2}=\left(\frac{p_2}{p_1}\right)^{\frac{1}{K}} \tag{A-2}$$

or, for a polytropic process:

$$\frac{Q_1}{Q_2}=\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} \qquad n=\frac{1n\left(\frac{p_2}{p_1}\right)}{1n\left(\frac{\rho_2}{\rho_1}\right)} \tag{A-3}$$

APPENDIX B

Due to very different ambient conditions (design 35°C, test 5°C), the engine is at 78 percent load during the test, although the design point was a full load point. It shows that, except under conditions even more extreme than in the example, the difference between the direct and indirect calculation is negligible. To calculate the uncertainties independently, as in the last line, will lead to exaggerated uncertainties.

- Calculated uncertainty for shaft power: 2.4 percent
- Calculated uncertainty, package heat rate (direct calculation): 2.0 percent
- Uncertainty, engine fuel consumption: 1.2 percent

• Difference in fuel consumption between test and design: 3.7 percent

• Uncertainty of the difference, relative to the design fuel consumption (3.7% * 1.2%) = 0.044 percent

• Calculated uncertainty, package heat rate (indirect calculation): $(2.0\%^2 + .044\%^2)^{1/2} = 2.0004$ percent

• Calculated uncertainty engine heat rate (independent calculation): $(2.4\%^2 + 1.2\%^2)^{1/2} = 2.7$ percent

APPENDIX C

If the temperature and the flow are measured without taking the recirculation into account, the absorbed gas power of the compressor will be calculated incorrectly (in the example, ideal gas is assumed for simplicity):

$$P_g = c_p(T_d - T_s)\, W_s \qquad P_{g,r} = c_p(T_d - T_{s,r})\, W_{s,r}$$

$$T_{s,r} > T_s \qquad W_{s,r} > W_s$$

$$\frac{\eta_r}{\eta} = \frac{\dfrac{T_d}{T_s} - 1}{\dfrac{T_d}{T_{s,r}} - 1} \tag{C-1}$$

In the above equations, the subscript r denotes the situation after the main flow and the recirculated flow have mixed, whereas no subscript indicates the values as they would be measured by devices upstream of the mixing point.

As shown above, the recirculation reduces the measured efficiency. This explains, why increased leakage of worn seals will reduce the efficiency of the compressor. Interestingly, it depends on the situation whether the measured gas power is higher or lower than the actual gas power, since the increase in actual flow is counteracted by the increase in actual suction temperature.

APPENDIX D

Straight lengths required for orifice plates, nozzles, and venturi nozzles [20]; minimum straight lengths required between various fittings located upstream or downstream of the primary device and the primary device itself.

• The unbracketed values are "zero additional uncertainty" values

• The bracketed values are "± 0.5 percent additional uncertainty" values

All straight lengths are expressed as multiples of the pipe diameter *D*. They will be measured from the upstream face of the primary device.

β	On upstream (inlet) side of the primary device							On downstream (outlet) side
	Single 90° bend or tee (flow from one branch only)	Two or more 90° bends in the same plane	Two or more 90° bends in different planes	Reducer (2 *D* to *D* over a length of 1,5 *D* to 3 *D*)	Expander (0,5 *D* to *D* over a length of 1 *D* to 2 *D*)	Globevalve fully open	Gate valve fully open	All fittings included in this table
≤0,20	10 (6)	14 (7)	34 (17)	5	16 (8)	18 (9)	12 (6)	4 (2)
0,25	10 (6)	14 (7)	34 (17)	5	16 (8)	18 (9)	12 (6)	4 (2)
0,30	10 (6)	16 (8)	34 (17)	5	16 (8)	18 (9)	12 (6)	5 (2,5)
0,35	12 (6)	16 (8)	36 (18)	5	16 (8)	18 (9)	12 (6)	5 (2,5)
0,40	14 (7)	18 (9)	36 (18)	5	16 (8)	20 (10)	12 (6)	6 (3)
0,45	14 (7)	18 (9)	38 (19)	5	17 (9)	20 (10)	12 (6)	6 (3)
0,50	14 (7)	20 (10)	40 (20)	6 (5)	18 (9)	22 (11)	12 (6)	6 (3)
0,55	16 (8)	22 (11)	44 (22)	8 (5)	20 (10)	24 (12)	14 (7)	6 (3)
0,60	18 (9)	26 (13)	48 (24)	9 (5)	22 (11)	26 (13)	14 (7)	7 (3,5)
0,65	22 (11)	32 (16)	54 (27)	11 (6)	25 (13)	28 (14)	16 (8)	7 (3,5)
0,70	28 (14)	36 (18)	62 (31)	14 (7)	30 (15)	32 (16)	20 (10)	7 (3,5)
0,75	36 (18)	42 (21)	70 (35)	22 (11)	38 (19)	36 (18)	24 (12)	8 (4)
0,80	46 (23)	50 (25)	80 (40)	30 (15)	54 (27)	44 (22)	30 (15)	8 (4)

	Fittings	Minimum upstream (inlet) straight length required
For all β values	Abrupt symmetrical reduction having a diameter ratio ≥ 0,5	30 (15)
	Thermometer pocket or well of diameter ≤ 0,03 *D*	5 (3)
	Thermometer pocket or well of diameter between 0,03 *D* and 0,13 *D*	20 (10)

REFERENCES

1. ASME PTC 10, "Compressors and Exhausters" (1979).
2. ASME PTC 19.1, "Measurement Uncertainties" (1985).
3. ASME PTC 22, "Gas Turbine Power Plants" (1985).
4. ISO 2314, "Gas Turbines—Acceptance Tests," Geneva, Switzerland (1989).
5. VDI 2045, "Acceptance and Performance Tests on Turbo Compressors and Displacement Compressors," Düsseldorf, Germany (1993).
6. VDI 2048, "Inaccuracies in Acceptance Test Measurements," Düsseldorf, Germany (1978).
7. Brown, R. N., "Fan Laws, the Use and Limits in Predicting Centrifugal Compressor Off Design Performance," *Proceedings of the Twentieth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, page 91 (1991).
8. Colby, C. M., "Hydraulic Performances Testing of Centrifugal Compressors—Hydrocarbon Versus Inert Gas," Proc. IMechE, paper C102/87 (1987).
9. Fozi, A. A., "Field Testing of Compressor Packages," Solar Turbomachinery Technology Seminar, Paper Number 98 (1995).
10. Schmidt, K-J., "*Experimentelle und theoretische Untersuchungen zum instationaeren Betriebsverhalten von Gasturbinentriebwerken*," VDI Reihe 7 (203), Düsseldorf, Germany (1992).
11. Beinecke, D. and Luedtke, K., "Die Auslegung von Turboverdichtern unter Beruecksichtigung des realen Gasverhaltens," VDI Berichte Number 487, pp. 271-279 (1983).
12. Reid, R. C., Prausnitz, J. M., and Sherwood, T. K., "*The Properties of Gases and Liquids*," New York, New York: McGraw-Hill Book Company (1977).
13. Doeblin, E. O., "*Measurement Systems*," New York, New York: McGraw-Hill Book Company (1966).
14. Brun, K., "Analysis of the Automotive Torque Converter Internal Flow Field," Ph.D. Dissertation, University of Virginia (1996).
15. Scarborough, J. B., "*Numerical Mathematical Analysis*," Baltimore, Maryland: Johns Hopkins University (1955).
16. Aust, N., "*Ein Verfahren zur digitalen Simulation instationaerer Vorgaenge in Verdichteranlagen*," Dissertation, University of the Federal Armed Forces, Hamburg, Germany (1988).
17. Cleveland, A., "Performance Testing of Gas Turbine Compressor Sets," ASME Paper No. 82-GT-199 (1982).
18. Schmitt, W. and Thomas, V., "Comparison of Test Measurements Taken on a Pipeline Compressor/ Gas Turbine Unit in the Workshop and at Site," ASME Paper No. 95-GT-125 (1995).
19. Boelcs, A. and Suter, P., "*Transsonische Turbomaschinen*," Karlsruhe, Germany: G. Braun (1986).
20. ISO 5167, "Measurement of Fluid Flow by Means of Orifice Plates, Nozzles and Venturi Tubes Inserted in Circular Cross-Section Conduits Running Full," Geneva, Switzerland (1980).
21. Meier, R. H. and Rhea, C. S., "Centrifugal Compressor Testing Experience and Practice," ASME Paper No. 82-GT-320 (1982).
22. McRoberts, I. W., "Performance Trials of Two Centrifugal Compressor Trains on an Offshore Platform," IMechE C48/84 (1984).

23. Valenti, M., "Determining Gas Meter Accuracy," Mechanical Engineering, *119*, (3), pp.102-106 (1997).

24. Klein, J. M. and Draughton, P. A., "Field Performance Testing of an Uprated Gas Reinjection Compressor/Turbine Train," ASME Paper No. 91-GT-48 (1991).

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